THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

94-GT-202

DESIGN AND DEVELOPMENT OF AN ADVANCED TWO-STAGE CENTRIFUGAL COMPRESSOR

D. L. Palmer and W. F. Waterman
AlliedSignal Engines
Light Helicopter Turbine Engine Company
Phoenix, Arizona

ABSTRACT

This paper describes the aero-mechanical design and development of a 3.3 kg/sec (7.3 lb/sec), 14:1 pressure ratio two-stage centrifugal compressor which is used in the T800-LHT-800 helicopter engine. The design employs highly nonradial, splitter bladed impellers with swept leading edges and compact vaned diffusers to achieve high performance in a small and robust configuration. The development effort quantified the effects of impeller diffusion and passive inducer shroud bleed on surge margin as well as the effects of impeller loading on tip clearance sensitivity and the impact of sand erosion and shroud roughness on performance. The developed compressor exceeded its performance objectives with a minimum of 23-percent surge margin without variable geometry. The compressor provides a high performance, rugged, low-cost configuration ideally suited for helicopter applications.

INTRODUCTION

Small turboshaft engines require high-pressure ratio, high-efficiency compressors to provide low engine fuel consumption. However, being in a low-power class, these engines must be inexpensive to manufacture and operate, requiring that the compressor be a simple, rugged design having low parts count and ease of manufacture. Helicopter applications, especially military, are subjected to high levels of sand ingestion and inlet distortion, requiring a durable configuration with good surge margin. The compressor for these engines has traditionally been accomplished by either an axial-centrifugal or a two-stage centrifugal configuration.

Particularly in flow sizes less than 4.5 kg/sec (10 lb/sec), engine and compressor sizing studies have indicated that the two-stage centrifugal compressor has several important advantages over an axial-centrifugal compressor:

- Good surge margin is achievable without use of variable geometry for steady-state operation, thereby reducing complexity, parts count, and manufacturing costs.
- Only two stages are required to achieve high pressure ratio (14:1 and greater) with good efficiency, again reducing complexity, parts count, and manufacturing costs.
- Greater blade thicknesses and rugged long chord blading improves foreign object damage (FOD) and sand erosion tolerance, thereby reducing overhaul costs.
- Reduced sensitivity to tip clearance can be achieved through low blade loading, thereby improving performance retention and reducing overhaul costs.
- High tolerance to inlet distortion is possible due to very low aspect ratio blading.

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
The Hague, Netherlands — June 13–16, 1994
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1994

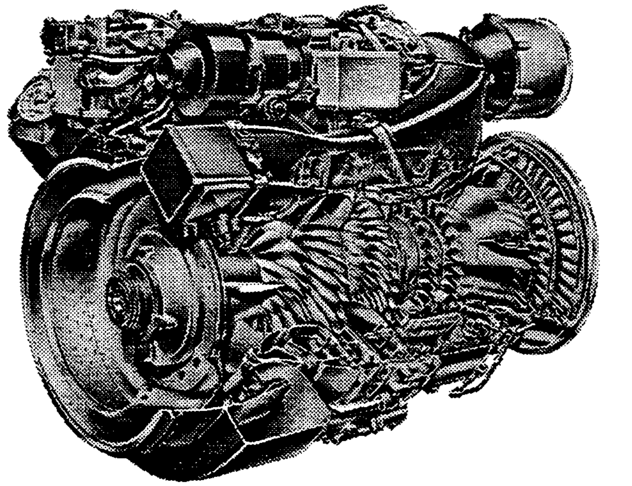


FIGURE 1. THE T800-LHT-800 ENGINE.

This paper describes the aero-mechanical design and development of a 3.3 kg/sec (7.3 lb/sec), 14:1 pressure ratio two-stage centrifugal compressor which is used in the T800-LHT-800 helicopter engine (Figure 1). The T800 is a 136 kg (300 lb), 895 kilowatt (1200 horsepower) class engine developed for the U.S. Army by the Light Helicopter Turbine Engine Company (LHTEC), a partnership between Allison Engine Company and AlliedSignal Engines.

As shown in Figure 1, this engine has a two-stage centrifugal compressor coupled to a two-stage gas-generator turbine with a two-stage freeshaft power turbine. The engine inlet is annular in nature and incorporates an integral inlet particle separator.

The compressor design employs highly non-radial, splitter bladed impellers with swept leading edges and compact vaned diffusers to achieve high performance in a small and robust configuration. The compressor development effort reported in this paper quantified the effects of impeller diffusion and passive inducer shroud bleed on surge margin as well as the effects of impeller loading on tip clearance sensitivity and the impact of sand erosion and shroud roughness on performance.

COMPRESSOR DESIGN

Cycle analyses of turboshaft engines indicated that the following compressor design objectives would provide the required power with the improved fuel consumption desired:

- Pressure ratio = 14:1
- Referred flow = 3.3 kg/sec (7.3 lb/sec)
- Adiabatic efficiency >0.80 (Total-Total including the inlet duct and rated to a combustor inlet Mach number of 0.15)
- Surge Margin >15% (Constant speed)

Preliminary design studies of the compressor indicated that a rotational speed (corrected to compressor inlet conditions) of 43,800 rpm was necessary to achieve the compressor overall diameter and weight objectives (with acceptable turbine stresses) for the turboshaft engine applications. This speed, combined with a first-stage inducer hub-tip ratio of 0.51 and an inlet absolute Mach number of 0.52, results in an inlet tip relative Mach number of 1.37. The relatively high hub-tip ratio was selected for three reasons:

- To allow the largest blade count practical with integrally machined impellers for high efficiency
- To minimize upstream duct curvatures and diffusion for low losses (the upstream duct is included in the compressor rating)
- To permit a short bearing span in a configuration incorporating front end drive

The impeller inlet absolute Mach number was selected to minimize upstream duct diffusion when matched to an inlet particle separator.

High impeller backward curvature (approximately 47 degrees) in both stages (Figure 2) was used to improve efficiency through reduced impeller loading and to improve surge margin by providing a steeper, and therefore more stable, energy addition characteristic. The high impeller exit blade angles required a coordinated aero-mechanical design effort, particularly for the first stage, to ensure acceptable blade aerodynamic loadings, stresses, and vibration characteristics.

High impeller blade count (14 main and 14 splitter blades) in the first stage was used to reduce blade loadings, thereby providing lower secondary flow losses and reduced sensitivity to tip clearance. To achieve high blade count while retaining low-cost manufacture, both impellers were designed with splitter blades. The blade

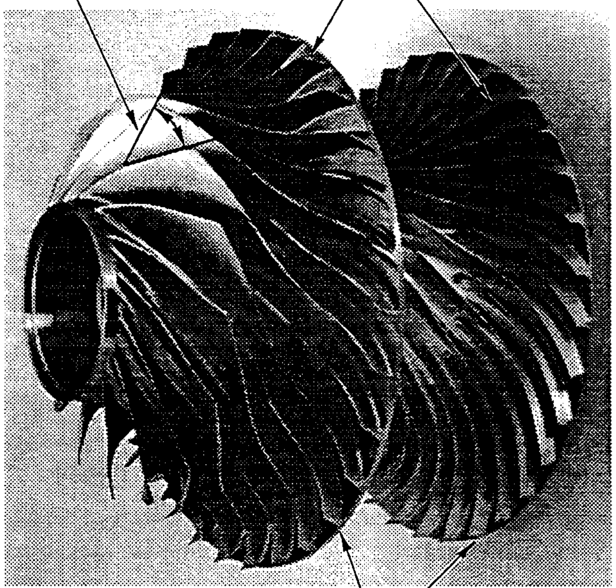


FIGURE 2. COMPRESSOR ROTATING GROUP ILLUSTRATES IMPELLER DESIGN PHILOSOPHY.

surfaces were generated by straight-line-elements, permitting the impellers to be integrally machined by flank milling with conical cutters on numerically controlled five-axis milling machines without encountering cutter tool interference with adjacent blading.

Because of the high inducer shroud Mach number on the first stage impeller, blade angle distributions were selected to minimize inducer tip turning upstream of the throat which, in conjunction with the high leading edge sweep, minimized shock losses and improved flow range.(Ref. 1) The high specific speed of this stage would normally result in a significant inducer tangential lean, which causes high blade stresses. However, by extending the leading edge hub ahead of the tip (i.e., adding leading edge sweep), these lean angles were significantly reduced, resulting in reduced stresses. The low inducer average lean angle is shown in a front view of the first-stage impeller (Figure 3) with a cross section cut perpendicular to the axis of rotation through the inducer tip of the main blade. The resulting impeller blade stresses (Figure 4) indicate a peak stress of 481 mpa (69.7 ksi) at the blade root, which is well below the allowable for forged titanium.

This design was accomplished prior to the availability of 3-D viscous analysis capabilities. Therefore, the impeller blade loadings were calculated using an axisymmetric radial equilibrium flow analysis for the meridional plane solution. Blade-to-blade analysis used the change in angular momentum through the impeller to calculate the surface relative Mach number distributions. Hub and tip loadings for the first-stage impeller are shown in Figure 5. These loadings indicate that the primary diffusion is taken early in the impeller to minimize high-suction-surface Mach numbers, shock losses, and resulting flow separation. Blade-to-blade loadings at the impeller exit are low as a result of using splitter blades. The splitter leading edges were positioned tangentially to minimize angle-of-attack losses as determined using an inviscid 3-D transonic flow analysis. As such, the splitters are not merely cut-off main blades.

To establish the choking flow capacity of the first-stage impeller and to verify the adequacy of the main and splitter blade designs, inviscid 3-D transonic flow analyses of the first-stage impeller were conducted using the method reported by Denton.(Ref. 2) Results of this analysis at conditions just out of choke are shown in Figure 6, which provides a view of the Mach number distributions on the suction surfaces of the main and splitter blades.

Two second-stage impeller designs were accomplished. The first design had 32 blades (16 main blades

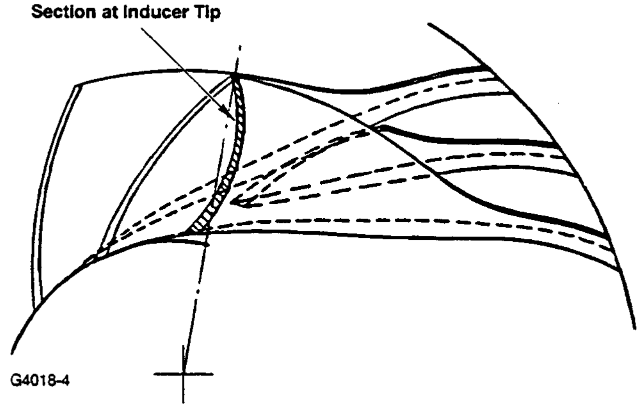


FIGURE 3. LEADING EDGE SWEEP MINIMIZES INDUCER LEAN ANGLES.

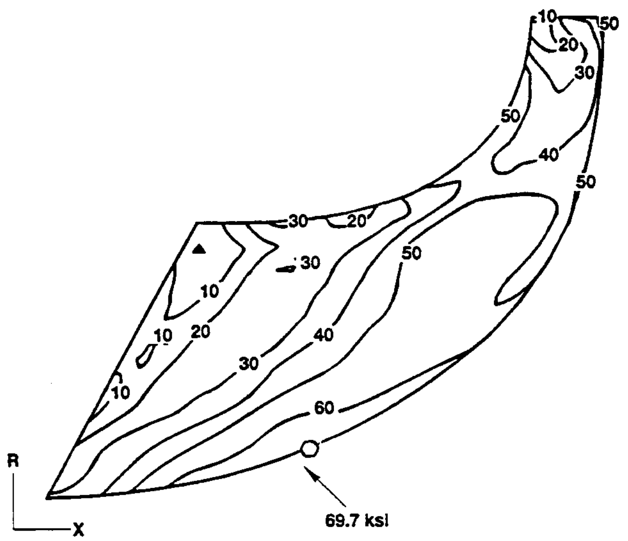


FIGURE 4. BLADE STRESSES ARE LOW DUE TO LOW INDUCER LEAN ANGLES.

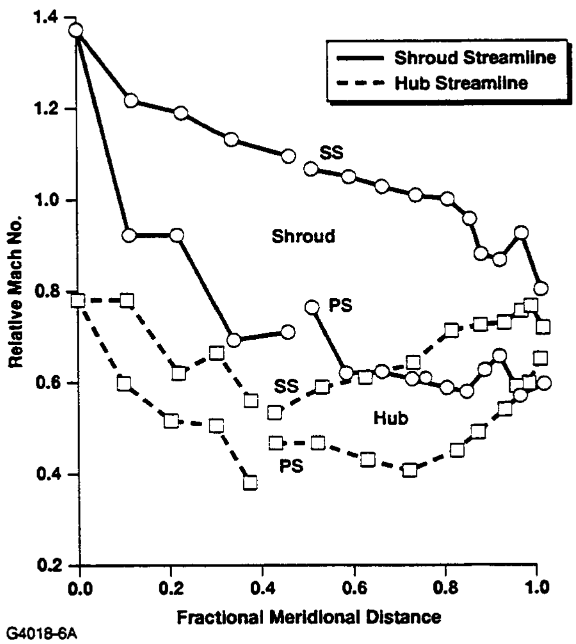


FIGURE 5. FIRST-STAGE IMPELLER LOADINGS INDICATE EARLY DIFFUSION AND NO OVER-VELOCITY.

and 16 splitters) with a nearly uniform meridional distribution of blade loading.

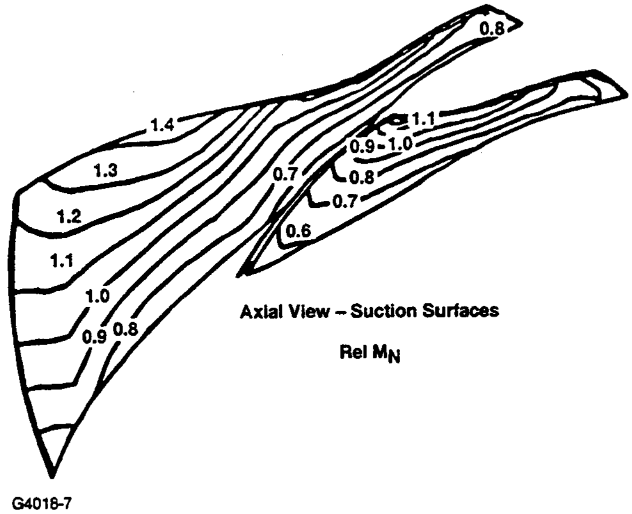


FIGURE 6. 3-D TRANSONIC ANALYSIS VERIFIES FLOW CAPACITY.

The major design goal of the other second-stage impeller was to significantly reduce impeller exit loading since low loadings reduce sensitivity to axial clearance. Low loadings were achieved by designing for 38 blades at the exit (19 main blades and 19 splitters) using similar techniques as described for the first-stage impeller. In addition, impeller blade angles were configured to accomplish nearly all of the required diffusion ahead of the splitter to further reduce impeller exit loadings. These loadings, from the radial equilibrium analysis, are shown in Figure 7 for both hub and shroud streamlines.

The compressor flowpath and diffusion system blading are shown in Figure 8. To reduce the first-stage overall diameter to minimize weight, the first-stage diffuser uses sheet metal vanes and splitters to maximize diffusion in a small radial envelope. The first-stage diffuser splitter vane is positioned tangentially to maximize the recovery of the two adjacent channels with uniform exit static pressure. The vanes were made of steel for FOD and sand erosion resistance and ease of manufacture.

The crossover duct between the two stages consists of a 180-degree annular bend followed by a radial inflow deswirl vane system which provides zero net swirl to the second-stage impeller. A small amount of bleed-air is taken off at the slot ahead of the second-stage impeller to simulate turbine cooling requirements. Acceleration bleed at this point is also provided for rapid transient operation.

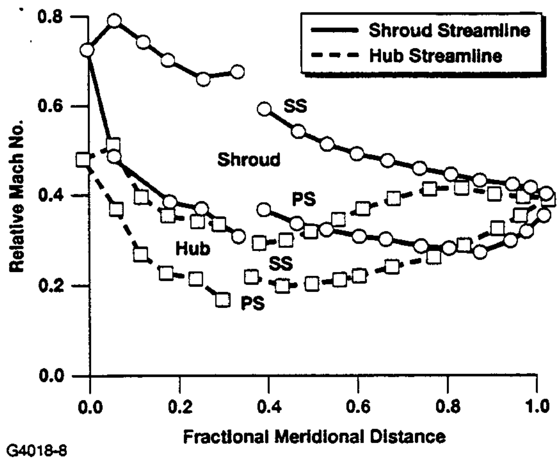


FIGURE 7. SECOND-STAGE IMPELLER HAS LOW EXIT LOADINGS FOR CLEARANCE INSENSITIVITY.

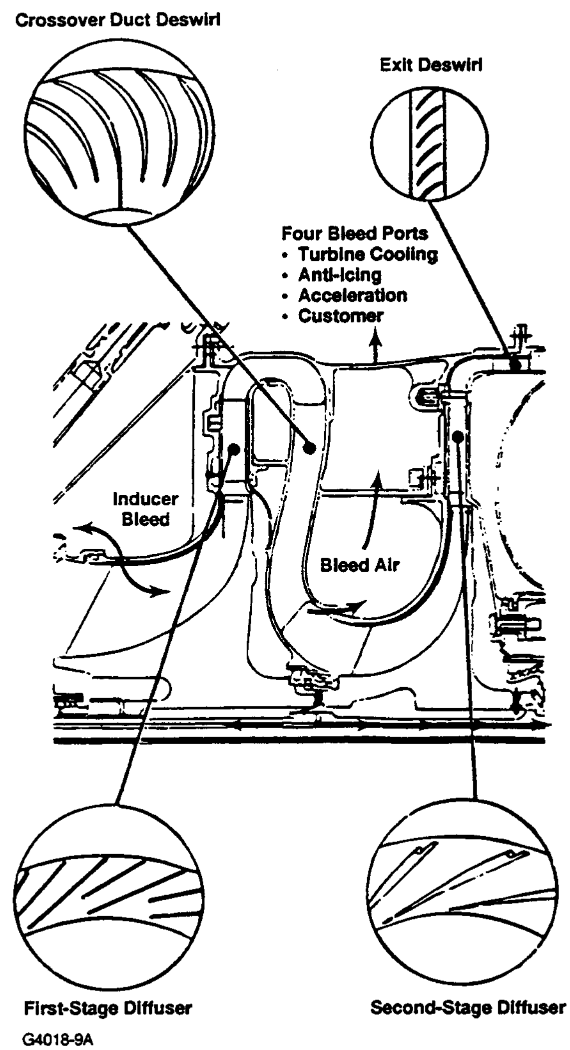


FIGURE 8. COMPRESSOR STATIONARY DIFFUSIONS SYSTEMS.

The second-stage diffuser is a conventional vane-island design with through-bolts. It achieves high recovery through a high divergence angle which is optimum for low-throat-aspect ratios. Splitter vanes were not used because sufficient radial envelope was available to provide the required recovery with a more conventional design. This diffuser is followed by a 90-degree annular bend and an axial deswirl row which reduces exit swirl to 30 degrees and exit Mach number to 0.15 entering the combustor.

The compressor design and performance rating includes an upstream transition duct and five service struts. During early development, this duct included the re-entry slots for a first-stage impeller passive inducer shroud bleed system similar to that reported by Chapman(Ref. 3) for single-stage centrifugal compressors. This system was developed for this two-stage centrifugal compressor during the rig test effort described below.

The initial passive inducer shroud bleed system, shown schematically in Figure 9, allows communication of the first-stage inducer flow with the inlet duct. When operating at part speed, the impeller pressure at the bleed port location is higher than duct pressure; therefore, outflow occurs naturally (without controls) from the impeller. This outflow, which returns to the duct at the re-entry slot, relieves inducer stall through natural low-flowing of the impeller and provides improvement in part-speed surge and efficiency (the recirculation occurs entirely within the compressor and losses are, therefore, properly accounted for in the compressor rating). At high speeds, the inducer approaches choke, resulting in a low pressure at the port location. This causes inflow to occur from the upstream duct, which reduces inducer choke losses with resultant increases in overspeed flow capacity and efficiency.

COMPRESSOR DEVELOPMENT

Development testing of this compressor was conducted using the test rig shown in Figure 9. This rig has a translating rotor group to control running axial tip clearances to match engine requirements. The rig can accept slip rings for strain gage testing, and a distortion generator can be attached upstream for pressure distortion testing.

Instrumentation for performance testing is also indicated on Figure 9. This instrumentation allows measurement of individual stage and overall performance, as well as providing diagnostic measurements of component static pressure distributions. Clearance is monitored with capacitive-type clearance probes. Testing at several axial clearance levels was conducted to establish compressor performance sensitivities. At high speeds, surge data was taken at approximately 0.25 mm (0.010 inch) increased clearance to avoid rubs. Previous testing has demonstrated that the spatial position of the compressor surge line on the map is not affected by this magnitude of clearance change.

Significant development testing was conducted to optimize the performance and surge margin of the compressor. During this development effort, a number of important factors affecting compressor performance and surge margin were tested. These factors included the effects of:

- First-stage impeller diffusion on surge margin
- Inducer shroud bleed system on surge margin
- Axial clearance on compressor performance
- Blade loading on axial clearance sensitivity
- Second-stage shroud roughness on compressor performance
- Sand erosion on compressor performance

Baseline Test

This first test of the as-designed hardware served as the baseline for the development program. The baseline test map is shown in Figure 10 along with a representative nominal engine operating line. Tip clearance testing during this build established clearance sensitivity for the original design second stage. This test showed good ef-

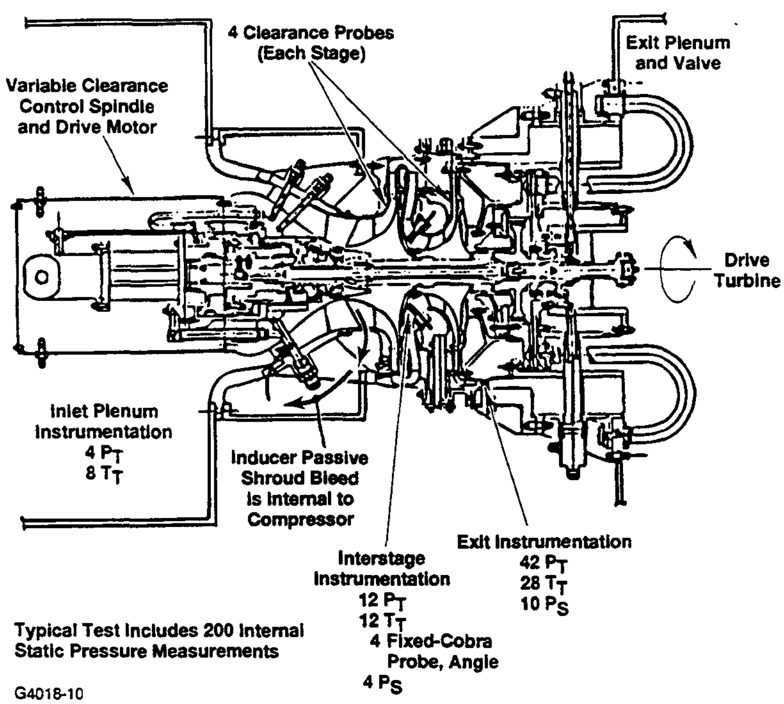


FIGURE 9. COMPRESSOR TEST RIG PROVIDED PERFORMANCE AND DIAGNOSTIC INSTRUMENTATION.

ficiency but was short of the objective surge margin at part speed.

Inducer Shroud Bleed

Since design point performance levels were close to objectives but surge margin was low, testing concentrated on surge margin improvement. Based on the work of Chapman and previous experience with passive inducer shroud bleed on two-stage centrifugal compressors, it was decided to incorporate such a concept for this two-stage centrifugal compressor. To facilitate changing the meridional position and area of the bleed ports along the first-stage impeller shroud, round holes through the shroud were used since they could readily be plugged or reduced in area as required.

This test used hardware from the baseline test with the first-stage inducer bleed system added. Tests were conducted with various hole positions and areas to optimize surge margin and performance. Inducer bleed flow was recirculated to the compressor inlet downstream of the inlet-measurement station, thereby accounting for all losses in the recirculation system. This testing showed that for an inducer bleed system using round bleed holes through the shroud, the best configuration consisted of a dual row of holes located in the throat region of the first-stage impeller.

Figure 11 shows the efficiency and surge margin test results for the dual row of holes compared to the baseline test. The following observations are made from this test:

- Surge margin above 80-percent speed was nearly doubled with the inducer bleed
- High-speed choke flow was increased
- Surge margin at 75-percent speed was reduced. Prior testing without the bleed system had shown that at this speed and below, the first-stage impeller inducer tip region is fully stalled, providing a stable internal recirculation. Under these conditions, adding a higher pressure drop external recirculation path actually reduces the recirculated flow and therefore the surge margin benefit.

Impeller Diffusion

This test was conducted to determine if reduced first-stage impeller diffusion, achieved by reducing the first-stage impeller exit and diffuser passage heights, could improve surge margin without significant loss in high-speed performance. Results of this testing, compared to the baseline, are shown in Figure 12. Again, improvements in surge margin above 80-percent speed were realized, although they were not as large as those

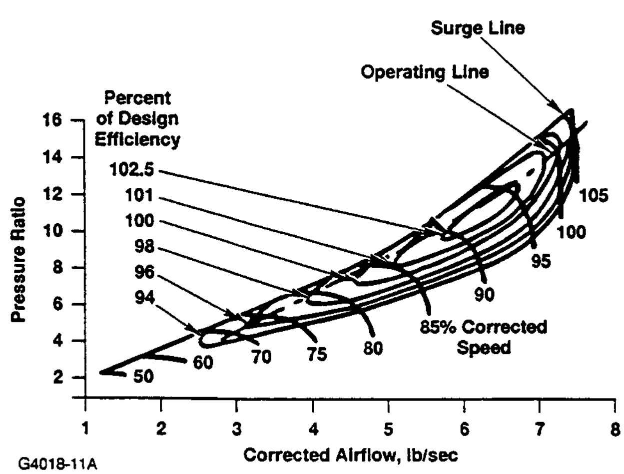


FIGURE 10. BASELINE COMPRESSOR TEST IS SHORT OF GOAL SURGE MARGIN AT PART SPEED.

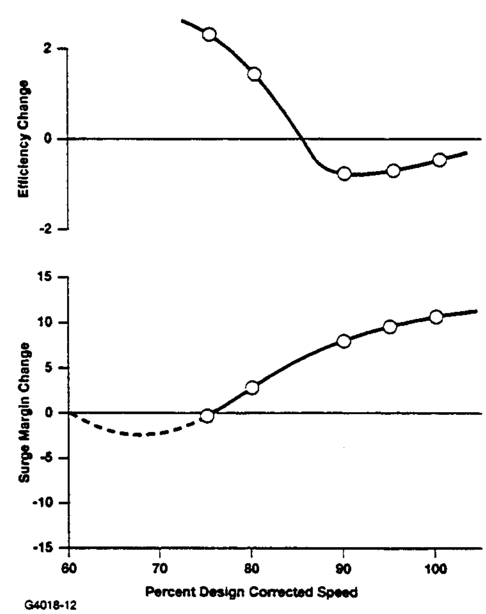


FIGURE 11. EARLY SHROUD BLEED CONFIGURATION SIGNIFICANTLY IMPROVED SURGE MARGIN OVER BASELINE.

achieved with the inducer bleed. Good high-speed performance was retained and no loss in surge margin occurred at 75-percent speed, as happened with the inducer bleed.

Shroud Roughness

Next, the redesigned second stage was introduced. This configuration is that described in the design section of this paper. At the same time, the second-stage shroud abradable coating material was changed which resulted in a surface roughness of approximately 9.0 µm (350 µin) compared to the previous configurations which had a roughness of approximately 1.8 µm (70 µin).

This test was conducted with the baseline first stage, the new design second stage, and the 9.0 µm (350 µin) roughness second-stage impeller shroud. Overall performance was disappointingly low and, as a result, a study of surface roughness effects in centrifugal stages was undertaken. The objective was to determine if the change in second-stage shroud roughness could account for the low performance observed on this test.

Using a pipe-flow analogy for a centrifugal stage, the characteristic length for similarity is the hydraulic diameter (which is nearly equal to the impeller exit tip width). On this basis, centrifugal compressor Reynolds number effects can be correlated, as shown in Figure 13, which shows the effects of surface roughness at high Reynolds numbers. The Reynolds number of the second-stage, operating in the two-stage compressor, is indicated on the figure ($Re_b = 3.5 \times 10^5$). As can be seen, the second stage lies in the hydraulically rough regime with a roughness of 9.0 µm (350 µin) and in the transition region between hydraulically rough and hydraulically smooth with a roughness of 1.8 µm (70 µin). The predicted change in second-stage efficiency for this change in roughness is 4.0 points, which converts to approximately 1.6 points in overall two-stage efficiency for the compressor.

As a result of the above analysis, the second-stage shroud coating material was restored to that used previously and the compressor retested. An improvement of over 2.0 points in overall efficiency was obtained at the design point.

Clearance Sensitivity

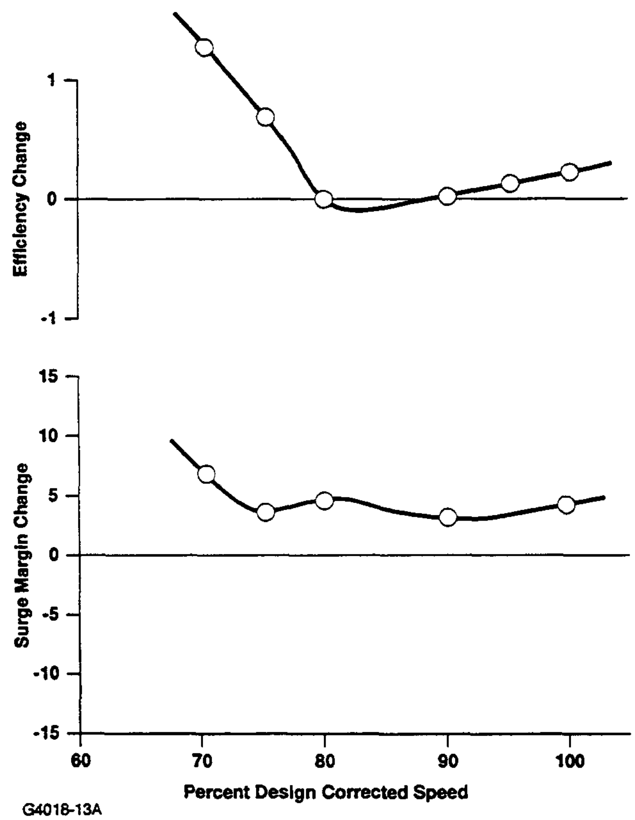


FIGURE 12. REDUCED FIRST-STAGE IMPELLER DIFFUSION IMPROVED PART-SPEED SURGE MARGIN OVER BASELINE.

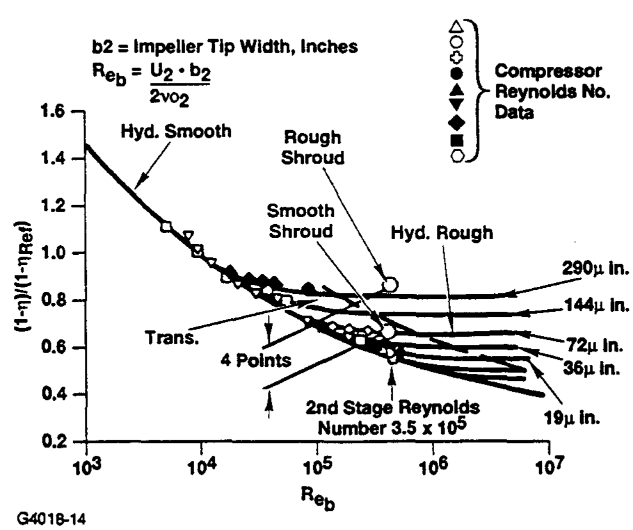


FIGURE 13. SINGLE-STAGE CENTRIFUGAL COMPRESSOR'S LOSS VERSUS ROUGHNESS FOR 0.3 INCH EXIT TIP WIDTH.

The sensitivity of the second stage to changes in impeller axial clearance was evaluated using the variable clearance feature of the compressor rig. Comparison of the clearance sensitivity of the original 16 main blade/splitter bladed second-stage impeller to the new design having 19 main blades/19 splitters is shown in Figure 14. Clearly, the reduced impeller loadings due to increased blade count and revised blade angle distribution has had a significant effect on the rate of efficiency loss with increasing axial tip clearance. The low clearance sensitivity allows the second stage to operate efficiently at a robust 6 percent axial clearance needed for the extremely rapid helicopter acceleration requirements. Comparison of these impellers is made in Figure 15.

Annular Shroud Slot Inducer Bleed

An annular slot inducer bleed configuration was also tested. The annular slot was located in a similar position on the first-stage impeller shroud as the dual row of round holes. The slot has approximately 2/3 the area of the round hole configuration due to the expected higher flow coefficient of the slot. A photograph of the slotted shroud is shown in Figure 16.

Comparative results of the two inducer bleed arrangements are shown in Figure 17. These results indicate that the annular bleed significantly improves surge margin over the dual hole configuration, particularly in the 70- to 85-percent speed region. The annular bleed design provided the compressor with a minimum of 13.5-percent surge margin over the entire steady-state operating region. However, detailed examination of the test data indicated that a modest rematching of the first-stage diffuser would improve performance, particularly in the important part speed (80 to 90 percent) region. This was accomplished by reducing the first-stage diffuser vane height by 3 percent, converging the vaneless space and reducing the diffuser throat and exit areas. This test resulted in the compressor achieving the efficiency objective and the minimum surge margin objective of 15 percent.

Overboard Venting of the Inducer Bleed

Additional testing was accomplished on a inducer bleed system with two changes; (1) the inducer bleed air was ducted outside of the engine, and (2) the impeller shroud slot area increased by 22 percent.

This configuration, used in the T800 qualification engines, yielded a large improvement in surge margin with a minimum of 23 percent and resulted in the map shown in Figure 18.

Strain Gage Testing

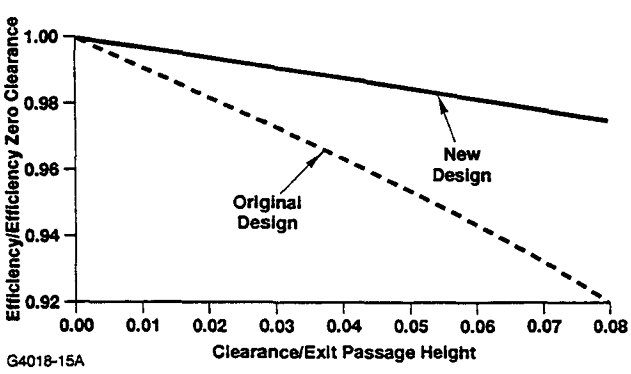


FIGURE 14. IMPROVED CLEARANCE SENSITIVITY OF THE SECOND STAGE COMPRESSOR BY REDUCED BLADE LOADING.

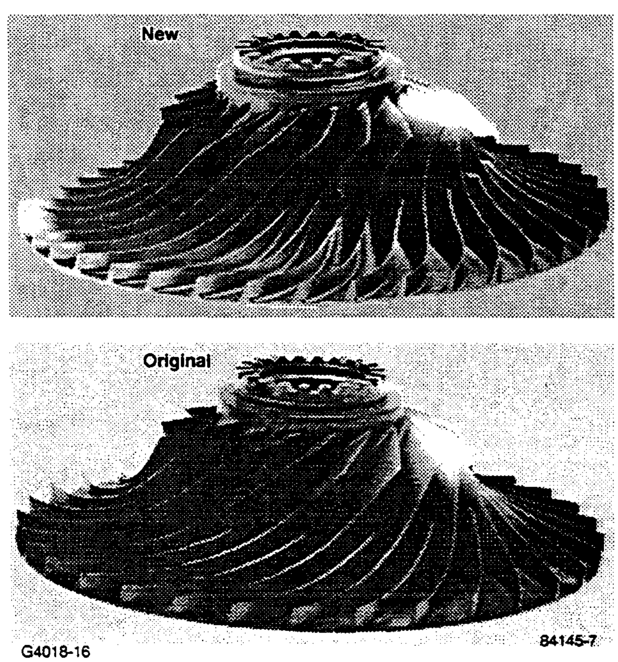
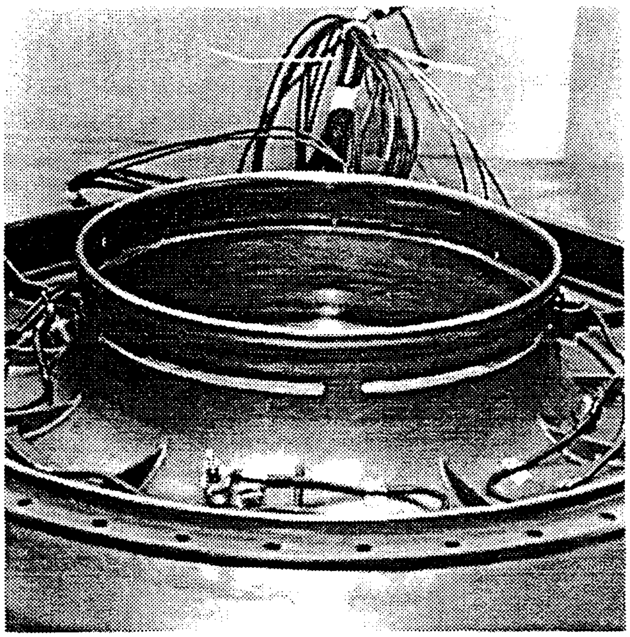


FIGURE 15. ORIGINAL AND NEW SECOND-STAGE IMPELLERS SHOW REVISED BLADE ANGLE DISTRIBUTIONS.



G4018-20 89721-6

FIGURE 16. INDUCER BLEED SLOTTED SHROUD TEST CONFIGURATION.

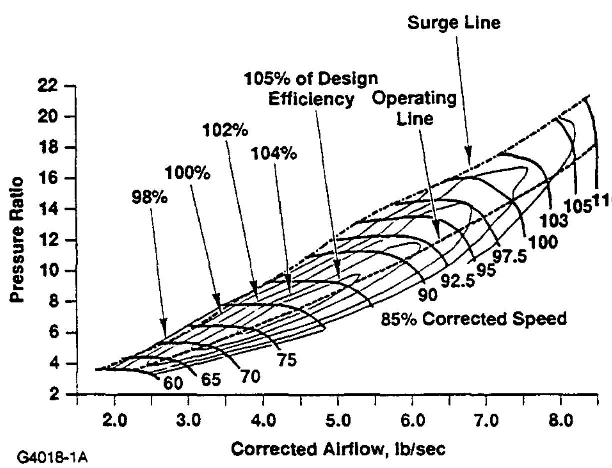


FIGURE 18. T800 FINAL COMPRESSOR MAP WITH OVERBOARD INDUCER BLEED ACHIEVES OBJECTIVES.

Strain gage testing was conducted with 1, 2, 3, and 4/rev circumferential distortion patterns and strain responses at critical locations on both the first- and second-stage impellers were measured. In no instance were the measured strains high enough to precipitate blade failures. Without distortion, strain responses were low enough that high-cycle-fatigue failures were not likely to

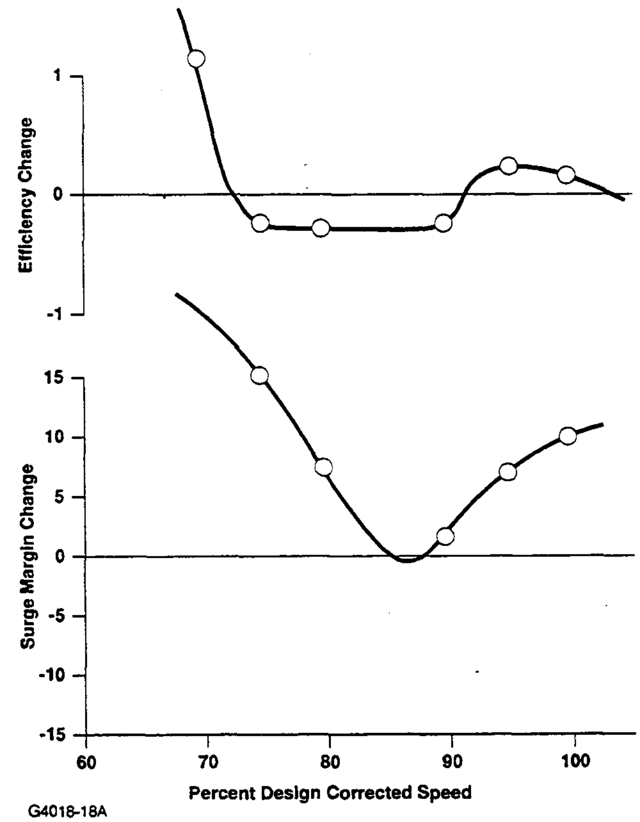


FIGURE 17. INDUCER BLEED SHROUD SLOT IMPROVED SURGE MARGIN OVER HOLE.

occur even with FOD impellers simulated by stress concentration factors of three.

Sand Erosion

The T800 engine was also subjected to a 50-hour C-spec sand ingestion test, an AC-fine sand test, and an AC-course sand test. The C-spec sand is generally considered to have the most potential for damage due to the high percentage of large particles. This test encompassed 50 hours of engine operation at maximum continuous rated power with sand contaminant introduced into the engine inlet at 53 mg per cubic meter of engine inlet air. Since the engine is equipped with an integral particle separator with a very high separation efficiency, most of the sand did not reach the compressor. However approximately 10.8 kg (2.39 pounds) of sand went through the engine core.

Engine performance deterioration was only about 5 percent horsepower loss at 25 hours and 8.3 percent at

50 hours. Diagnostic analysis of the pre- and post-test data indicated the compressor efficiency was decreased 1.1 to 1.3 percent over the range of 85 to 100 percent speed and was the major cause of the engine performance loss.

Post-test inspection of the compressor hardware showed:

- The leading edges of the first stage impeller blades were eroded back about 0.50 to 0.85 mm (0.020 - 0.034 in), as shown in Figure 19, with the higher value near the hub
- A slight channel 0.38 mm (0.015 in) deep on the pressure side of each blade and splitter at the exducer of the first-stage impeller
- An increase in roughness on the first-stage shroud abradable coating
- No visual signs of second-stage impeller leading edge erosion were noted
- The second-stage shroud had a polished appearance
- Only minor evidence of erosion was noted on the first- and second-stage diffusers and deswirl assembly

CONCLUSIONS

The design and development work described in this paper has shown that a high-pressure-ratio, two-stage centrifugal compressor is capable of high performance and that good surge margin and distortion tolerance, necessary for helicopter applications, can be achieved without resorting to variable geometry.

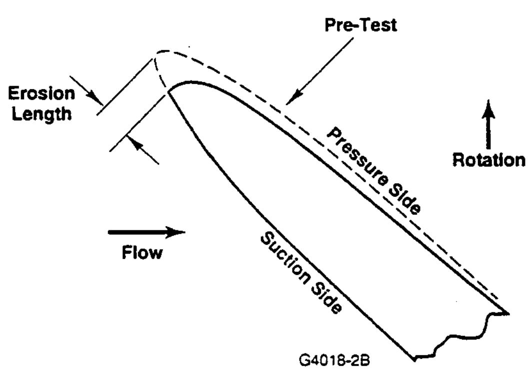


FIGURE 19. FIRST-STAGE IMPELLER LEADING EDGE SAND EROSION.

The passive inducer bleed concept was shown to improve surge margin, and part-speed and overspeed performance.. In addition, the concept also improved the already exceptional pressure distortion tolerance of this compressor configuration as described by Cousins, et al.(Ref. 4,5)

Development testing also showed that:

- First-stage impeller diffusion levels impact surge margin
- Shroud roughness of high-pressure centrifugal stages has a significant effect on performance
- Tip clearance sensitivity can be reduced with lower blade loadings

The developed T800 compressor provides a fully defined, high performance, and robust configuration ideally suited for helicopter applications.

REFERENCES

1. A. J. Wennerstrom and S. L. Puterbaugh, ''*A Three-Dimensional Model for the Prediction of Shock Losses in Compressor Blade Rows,*'' ASME Paper 83-GT-216, 1983.
2. John Denton, ''*An Improved Time Marching Method for Turbomachinery Flow Calculation,*'' ASME Paper 82-GT-239, 1982.
3. D. C. Chapman, ''*Model 250-C30/C281 Compressor Development, AGAED Compressor Proceedings No. 282, Centrifugal Compressors, Flow Phenomenon, and Performance,*'' May 1980.
4. W. T. Cousins, K. K. Dalton, T. T. Andersen, and G. A. Bobula, ''*Pressure and Temperature Distortion Testing of a Two-stage Centrifugal Compressor,*'' ASME Paper 93-GT-118, 1993.
5. W. T., Cousins, R. E. Miller, and K. K. Dalton, 1991, ''*Distortion Tolerance of the T800-LHT-800 Turboshaft Engine,*'' Proceedings, American Helicopter Society 47th Annual Forum and Technology Display, Vol. 2 (A92-14326 03-01) pp 1147-1155, 1992.